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Use of noise and vibration signal for detection and monitoring of cavitation in kinetic pumps

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Cavitation in kinetic pumps reduces delivery head and efficiency of a pump. This causes mechanical damage and increase of vibrations and noise. Therefore, it is important to detect inception and development of cavitation phenomenon in the pump. This paper deals with signals of vibrations and noise, which will be used for detection and monitoring of cavitation in kinetic pumps, and also to prevent the effect of cavitation in the pump and pumping system. When the cavitation is increasing, the flowing conditions are changing, which leads to an increase of vibrations of the pump and emitted noise in the surroundings. Because vibrations and noise are transferred from the pump through its casing, the signal is non-uniformly distorted due to transfer losses and structure of the casing surfaces. Noise and vibrations are increasing steadily, but in some specific frequency ranges the signal is more pronounced than in other part of the spectrum. Experimental results have shown that when the cavitation is fully developed, the measured signals at a characteristic frequency or range of frequencies start to decrease. This characteristic frequency or band of frequency is discussed in this paper, and also comparison between theoretical expectations and measurement results was performed.

1 Introduction

Despite well known problems about cavitation phenomenon in pumping systems, there are still many questions, waiting for researchers. Cavitation causes lowering performance and efficiency, physical damage and also increased vibrations and noise. It occurs when static pressure at some point within a pump falls below vapour pressure of the fluid at the prevailing temperature. The fluid then starts to vaporize and many gaseous bubbles arise. When the cavitation is increasing, the number and the size of bubbles are also increasing. Bubbles travel in the fluid through the pump and somewhere they reach an area with higher pressure, where they rapidly collapse. Forces, generated during the collapse, can have very destructive consequences. If the implosion of the bubble occurs near solid wall, shock wave reaches the surface and erodes it. Therefore the pump surfaces are damaged which causes non-ideal flowing conditions and after a long period of time the pump is useless. It is obvious that the cavitation phenomenon should be observed and prevented as soon as possible.

There is a few different methods for cavitation monitoring that can be used to prevent destructive consequences of the cavitation in the pumping systems [1]; determination of the net positive suction head (NPSH) at a constant speed and flow rate, visualization of the inlet flow at the impeller, paint erosion on impeller blades, static pressure measurement in the flow, vibration measurement of the pump structure and acoustic measurement in the pump. Acoustic measurement is well known by measuring ultrasound inside of the pump, but not so well by measuring audible noise in proximity of the pump.

In this paper, detection of cavitation is investigated by monitoring noise and vibrations, both measured outside of the pump and in a frequency band between 20 Hz and 20 kHz. Despite many losses in signal transfer between a source and a microphone or an accelerometer, it is possible to determine the onset of cavitation in a pump. In general, noise and vibration levels gradually increase until the cavitation is fully developed and then lower when gaseous phase is increased further. Therefore the correlation exists between the point of fully developed cavitation and the maximum of noise and vibration level. This allows the system to know when special steps have to be done to prevent the pump from operating in cavitating conditions.

Sound measurement in the proximity of the pump is limited not only by signal losses, but also by strong noise from the surroundings. But still this method is very useful because its simplicity. Measuring noise and vibration outside of the pump does not interfere with the fluid inside. Vibration measurement is less dependent on the surrounding noise, therefore in combination with the noise measurement it forms pretty reliable method for cavitation monitoring.

2 Cavitation phenomenon and its effects

While operating, a pump has to overcome a pressure difference, which arises by pipe resistance, height difference between pump and fluid level and eventually negative pressure above the fluid level. In some circumstances the given pressure difference is too high for the pump to be capable of pumping the fluid, which means that the NPSH value is insufficient. On suction side the pressure becomes very low and it can reach vapour pressure of the liquid. It happens first at the impeller eye, where the pressure is the lowest. The higher the flow rate, the sooner the cavitation arises, because the flow velocity is higher and consequently the pressure is lower.

When the pressure in the fluid drops to the vapour pressure of the fluid and lower, the bubbles filled with air or vapour start to rise. There are two bubble generating mechanisms [2]. First, there is always some dissolved gas in the fluid. When the pressure drops, conditions change and dissolved gas comes out of the fluid as a gas bubble. This is called gaseous cavitation. And second, when static pressure drops below vapour pressure, vaporisation occurs and bubbles start to grow. This is called vaporous cavitation. Fluid volume is increased significantly when vaporized. In case of a fully developed cavitation, the water at 21°C, for example, increases in volume about 54.000 times [3]. Theoretically, it can be calculated the size of particular bubble by using of the generalized Rayleigh-Plesset equation for bubble dynamics [4],

$$\frac{p_B(t)-p_\infty(t)}{\rho_L} = R\frac{d^2R}{dt^2}+\frac{3}{2}\left(\frac{dR}{dt}\right)^2+\frac{4\nu_L}{R}\frac{dR}{dt}+\frac{2S}{\rho_L R} \qquad (1)$$

where ρ_L is fluid density, R bubble radius, t time, ν_L kinematic viscosity of the fluid and S surface tension. Values of the pressure in the fluid $p_\infty(t)$ and in the bubble $p_B(t)$ are known. It was shown by Blake (1949) and

Neppiras and Noltingk (1951) that critical radius exists. The bubble is stable if his radius is smaller than critical radius, and it will grow fast, if radius is bigger. This is often referred to as Blake critical radius [4],

$$R_C = \left[\frac{9km_G T_B K_G}{8\pi S}\right]^{\frac{1}{2}} \tag{2}$$

where k is a polytrophic constant, m_G mass of gas in bubble, T_B bubble temperature and K_G gas constant. Critical pressure, derived from critical radius, is known as Blake threshold pressure,

$$p_{\infty c} = p_V - \frac{4S}{3}\left[\frac{8\pi S}{9km_G T_B K_G}\right]^{\frac{1}{2}} \tag{3}$$

where p_V is a vapour pressure. This means that pressure below critical pressure will cause the bubble to grow fast.

If the pressure is reduced further, the bubbles are growing and travelling with the fluid through the pump. When they reach the higher pressure area, they implode rapidly. This effect causes relaxation of very huge amount of concentrated energy, which can be very destructive. If the bubble implodes close to the solid surface (on impeller blade or spiral casing) a shock wave appears, which hits the surface and damages it. This continuous process leads to carrying away the material, which results in increased surface roughness and eventually perforated impeller. This causes improper flowing conditions and worse efficiency, and also increased noise and vibrations. Therefore the cavitation phenomenon in a pump is unacceptable and should not be tolerated.

3 Sources of noise and vibrations in a pump

Many different mechanisms govern noise and vibrations in a pump. They can be divided into those caused hydro dynamically by the flow perturbances in the pump, mechanically caused by unbalanced rotating masses, friction and vibration caused in bearings, friction in seals and structural caused by vibration of the structure (pump housing). The noise caused by driving electric motor is out of scope of this paper.

The lowest levels of noise are expected in a region of optimal flow rate. If the operating point is somewhere outside of this range, noise and vibrations are increased due to instability and pressure pulsations, caused by incorrect impeller inflow and appearance of flow recirculation at partial flow rates. The last happens at flow rates, usually below 65% of the optimal flow rate [5]. Pressure pulsations in general increase with the square of the velocity and have their minimum near the best efficiency point. Level of pressure pulsations depends on many parameters: geometry of impeller blades, diffuser (with or without vanes) and its geometry, velocity difference between suction and pressure side of the blade, impeller and diffuser flow passages, impeller inlet flow conditions, clearance between impeller blades and diffuser vanes or volute tongue and other geometry and system conditions [6].

Important source of noise and vibrations is cavitation, which is the main subject of this paper. Implosions of bubbles near pump surfaces cause many pressure shocks. These kinds of shocks depend on (1) size and number of bubbles and (2) type and material of impeller. Large bubbles signify stronger cavitation and lower frequency range due to large volume of the bubbles and their strong damping. Great number of bubbles signifies also stronger cavitation and more broadband frequency range due to large surface, on which they implode. If the impeller is open or semi-open type, stronger noise on the outer side of the pump is expected. If it is closed type, pitting shocks are transferred outward mostly through the bearings and less directly through the casing. This means greater damping and greater scattering of noise and vibrations in the pump.

Noise from bubble implosions at fully developed cavitation is heard by human ear and is often indication of cavitation in the pump. But increased noise and vibrations occur much sooner than they are heard. Growth of gas bubbles causes local blockages of a fluid, therefore the velocity of a fluid has to increase in order to maintain the same flow rate. This causes non-ideal flowing conditions, non-ideal inflow on impeller blades and because of non-homogeneous fluid density (two-phase flow) local recirculation form, which leads to increased noise and higher level of vibrations. Regarding small pump dimensions in our case, more activity can be expected in higher frequency range. Such dimensions prevent the structure to vibrate at frequencies with wavelength, significantly longer than the greatest pump dimension.

4 Acoustical detection of cavitation

Methods for acoustical detection of cavitation are known and described for a long time (for example [2, 7, 8, 9]). But almost every study is limited on investigation of higher frequencies, above 20 kHz, in ultra sound. Measurements are often performed inside the pump, at places where cavitation is more likely to occur (for example, at the impeller eye), using hydrophone or accelerometer. Low frequency noise (below 20 kHz) is usually filtered out because it contains too much noise in correlation with rotation (blade passing frequency, rotating frequency and their harmonics). Characteristically noise is increased when delivery head (H) drops. It is also known that a correlation exists between pressure level amplitude and cavitation erosion intensity, which increases with increasing noise [8].

When cavitation is fully developed and a pump contains a great amount of gas phase, noise and vibrations start to decrease. This is known occurrence and is well explained in literature [10], and is also confirmed with measurements. Reason for this is in absorption of pressure energy by present bubbles. At the same time they absorb pump energy, therefore the delivery head drops quickly.

Although researches in the past did not give special attention to audible sound for cavitation detection, this method offers some important advantages; measuring is simpler and there is no need to modify the pump for the transducer to build in. Despite noise from the surroundings and the noise in correlation with rotation, this method can be used as reliable indication of cavitation occurrence in a pump.

Audible sound was already used for cavitation detection and measured results have shown that this method has a good potential [1]. It was shown that in case of cavitation the noise increases in broadband and also at specific discrete frequencies.

5 Measuring procedure

Measurements were performed on a special test stand, according to the ISO 3555 (quality level B), and it is shown in Fig.1.

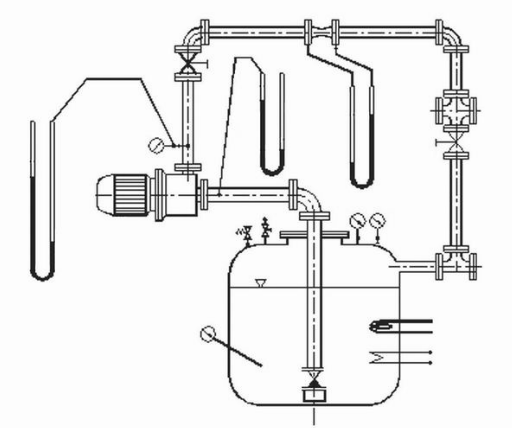

Fig.1 Test stand for pump measurements.

Two centrifugal pumps were tested; the first has 6 blades closed impeller and is made of metal alloy (pump A), and the second has also 6 blades, but semi-open impeller and is made of plastic material (pump B). The test stand is made in a closed loop. Water is pumped from a pressure vessel in which pressure is adjusted to get lower NPSH value. Then the water is directed upwards through the pump and a valve for flow rate adjustment, and back to the vessel. Vibrations were measured mainly on the pump casing, while noise was measured in points around the pump, mainly in a near-field (from 1 to 15 cm from a volute casing). It was found out that measuring noise in a near-field is better choice because in that case more useful signal is caught by the microphone. In a certain distance from the pump, useful noise is mixed with the surrounding noise which makes the measurement worse. The signal was amplified by measuring amplifier B&K 2636 and led to the computer, where it was analysed with frequency analysis software. At the same time the NPSH value and the delivery head were measured to track the cavitation development and to determine the point of fully developed cavitation (3% drop of delivery head). Cavitation development was observed at different flow rates. Through the particular measurement the flow rate was maintained constant, except when the cavitation was developed to the point where it was not possible to adjust the flow rate any more.

6 Non-cavitating conditions

According to the fact that pumps A and B were made of quite different material, certain differences were expected in measured frequency spectra. Pump A, made of metal alloy, is more rigid and particular frequencies are more pronounced than in case of pump B, which is made of plastic.

Comparison of noise and vibration spectra of both pumps is shown in Figs. 2 and 3. Both pumps show influence of rotating frequency (RF) at 49 Hz, blade passing frequency (BPF) at 295 Hz and their harmonics. But in case of pump A, RF and its harmonics are more pronounced, while pump B pronounces mainly BPF and its harmonics, which is best shown in vibration spectrum in Fig.2. This could be due to different impeller types of the pumps. A closed impeller offers less contact between blades and surrounding elements, therefore only fundamental frequency is emphasized.

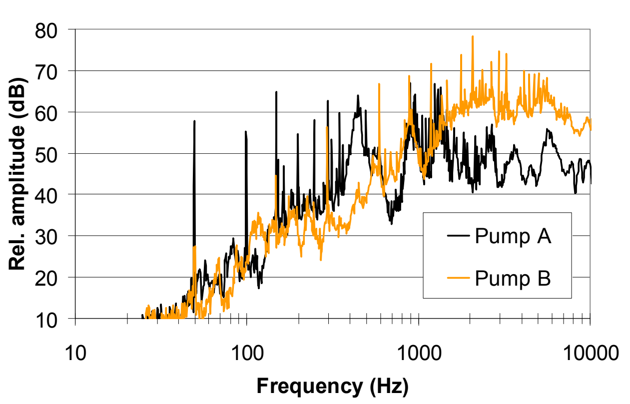

Fig.2 Vibration spectra of both pumps.

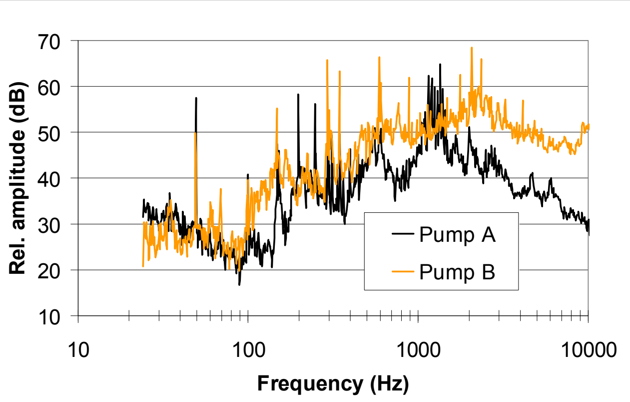

Fig.3 Noise spectra of both pumps.

On the other hand, a semi-open impeller has one side open to the housing, which means that every single blade has more influence on emitted nose and vibrations due to vortices appearing in axial clearances between rotor and housing, which are occurring due to the pressure difference between the pressure and suction sides of the rotor blades. In addition, noise generated within the rotor is directly transmitted to the housing and excites it in vibration and consequently generates structure-borne noise.

7 Cavitating conditions

A diagram of sound pressure level in audible range in dependence of NPSH value shows similar results to those of measured in ultrasound (in papers mentioned before). Figs. 4 and 5 show the total delivery head (*H*) and vibration level (*La*) for pumps A and B, when lowering NPSH value. The results are stated for the optimal flow rate (7 l/s) and narrow frequency band around 1600 Hz for the pump A or at the discrete frequency 148 Hz for the pump B. The corresponding diagrams for sound pressure levels are very similar to those presented in Figs. 4 and 5 , therefore they are not presented here.

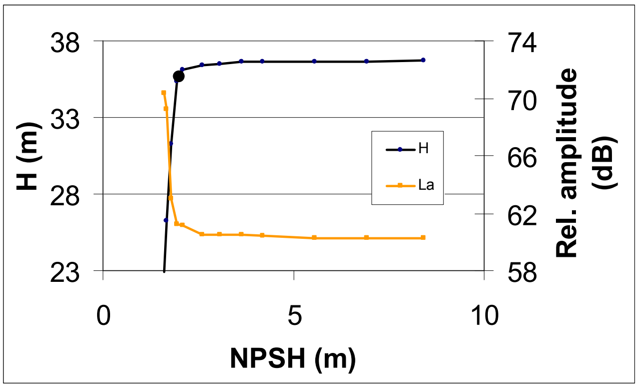

Fig.4 Vibration signal when lowering NPSH, pump A.

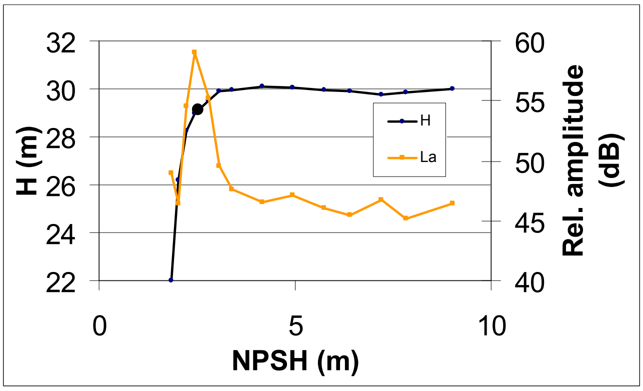

Fig.5 Vibration signal when lowering NPSH, pump B.

In Figs. 4 and 5, the big black dots denote the critical point, where total delivery head decreases for 3%, which means that there is fully developed cavitation. Vibration (or noise) signals of both pumps are evidently significant for detection of cavitation phenomenon. Noise and vibration levels at the critical point start to increase steeply (pump A) or reach the maximum level and then start to decrease steeply (pump B).

Figs. 6 and 7 show measured drop of delivery head when the NPSH value is decreasing, at different flow rates. At higher NPSH values, the delivery head is constant, but when cavitation starts to progress stronger, it drops down. Black dots in Figs. 6 and 7 represent critical value of the NPSH for particular flow rate, from 2 to 11,4 l/s, and the constant speed of rotation.

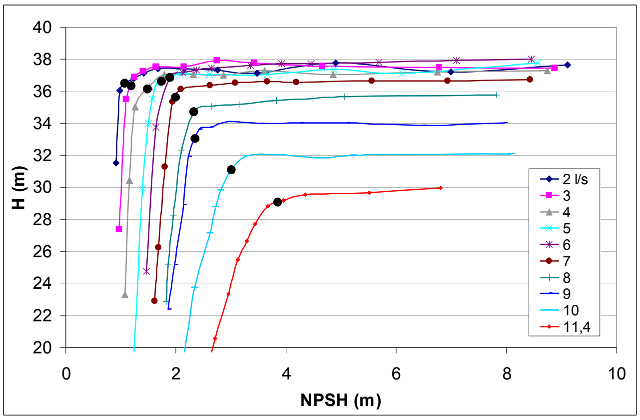

Fig.6 NPSH characteristics of the pump A.

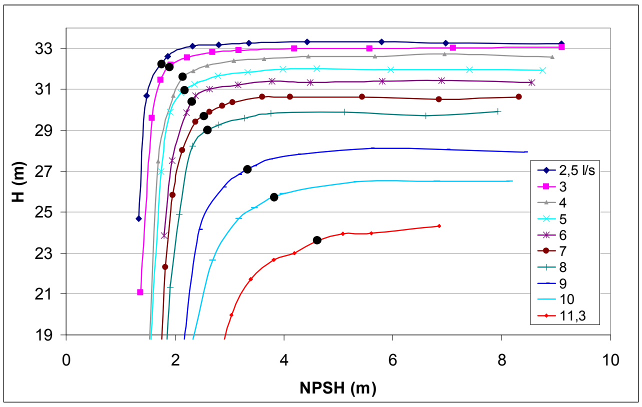

Fig.7 NPSH characteristics of the pump B.

The more the NPSH value decreases, the higher noise and vibration levels are. Many gas bubbles form and then grow with the development of cavitation. Therefore cavitation clouds form, flow passages reduce and consequently local velocities in the pump increase. This leads to increase of the broadband noise and vibrations. It is well known that cavitation appears much sooner than drop of delivery head is appeared [7], but in this stage it is not dangerous for damage of the pump.

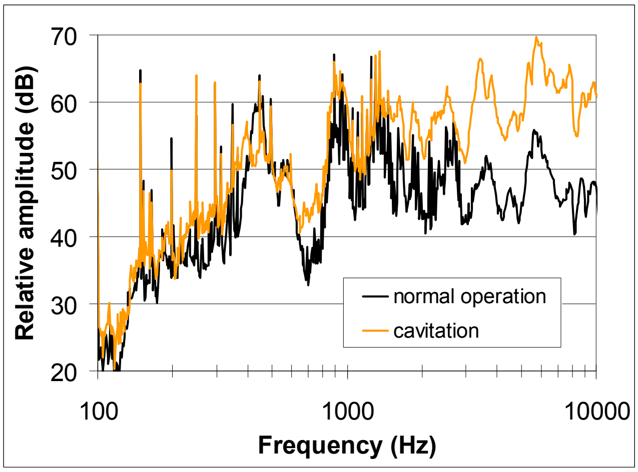

Fig.8 Vibration spectra of pump A, with and without cavitation.

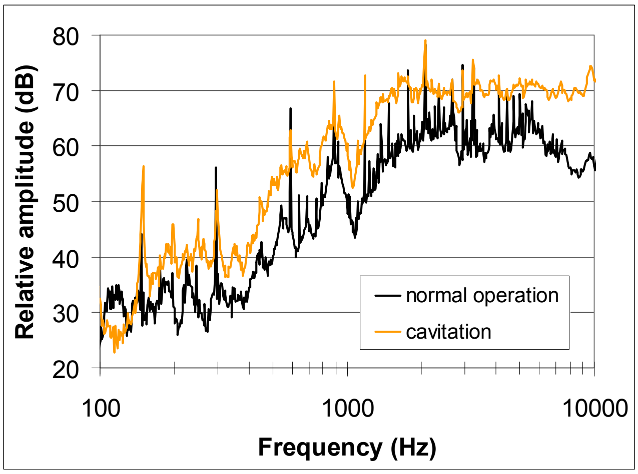

Fig.9 Vibration spectra of pump B, with and without cavitation.

In Figs. 8 and 9 the comparisons between spectra of the vibration levels of pumps A and B with and without cavitation phenomenon are presented, at the optimal flow rate (7 l/s). Vibrations are increasing through the complete frequency range, but at lower frequencies the spectra are mixed with strong harmonics of RF and BPF. At higher frequencies (above 3 kHz at the pump A and above 6 kHz at the pump B) harmonics of the RF and BPF are not so strong any more and the signal contains mainly broadband noise. Therefore this higher frequency range can serve for cavitation detection, because noise and vibration levels with and without cavitation differ for about 10 dB, almost equally through wide frequency range and for all measured flow rates.

But when cavitation in the pump occurs, additional discrete frequency tones or narrow frequency ranges appear in the noise and vibration spectra. Significant for the pump A is narrow frequency range around 1600 Hz, which steeply increases when cavitation start to develop stronger, at different operating conditions (flow rates). This frequency range can be used as reliable indicator for monitoring the cavitation phenomena.

Significant for the pump B is a discrete frequency at a 148 Hz, at which noise and vibration levels increase for about 15 dB and more. This frequency corresponds to first sub harmonic of the BPF. Important research about this phenomenon was already performed years ago with noise measurements [1] and with this paper it was confirmed with vibration measurements. It is also known that strong periodical pressure fluctuations cause sub harmonics, for example in a turbulent flow [11]. In case of a pump, the following explanation from Guelich and Bolleter [6] could be possible. If the local pressure at some point in a system containing a compressible fluid is subject to a sudden variation, a pressure wave travels through the system with the speed of sound of the fluid. Let it be a small sphere in the “hydrodynamic far field”. If the sphere is moved periodically around a zero-position, water is displaced which causes periodic variations in local velocity and pressure. A stationary object introduced into the near field acts itself as an additional source of noise. The acoustic energy radiated from this object depends on the velocity fluctuations in the near field. Therefore the pressure fluctuations created from this secondary noise source can exceed strongly the noise of the primary source.

A significant frequency of pump B has also higher harmonics, which arise when cavitation is fully developed. The most visible one is the second harmonic at a 444 Hz, but is not as reliable as 148 Hz and does not appear at all measured flow rates. At first harmonic (296 Hz), increase in noise and vibration level is not noticeable because influence of BPF is too strong and it drowns out any other signal.

8 Conclusions

Both, noise and vibration signals can be used for detection of cavitation phenomenon. Which signal is more appropriate in a certain case depends on surrounding noise. If the surrounding noise is very high and it is not easy to exclude it, vibration measurement is more appropriate. If not, noise measurement can be used since this method is simpler.

Each pump has specific characteristics, which depends on the composing material, type of impeller, form and dimensions, and therefore has different spectra and level of noise and vibration, and so different characteristic discrete frequency. It means that their responses are different and they can not be directly compared. Detection method should be determined for each pump individually.

Noise or vibration signal differs, if a pump operates at the optimal or at off-design flow rates. With method, presented in this paper, cavitation phenomenon can be detected in a range of usually used flow rates. Differences in noise and vibration levels between cavitating and non-cavitating conditions are above 10 dB, somewhere also above 15 dB for discrete frequencies and about 10 dB for broadband noise in higher frequency range.

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